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Characterization of a solar-powered fluidyne test bed



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ABSTRACT

Liquid piston Stirling engines (sometimes termed “fluidyne” engines) have been studied extensively and applied in a variety of energy conversion applications. They are attractive for low capital costs and simplicity of construction. In addition, their operation as external combustion engines allows for flexibility in primary energy sources which is a distinct advantage when a low-cost or free source of heat can be paired with their minimal construction costs. Disadvantages of these devices include relatively low efficiency and low power density. A solar-powered fluidyne test bed was constructed and tested extensively. The test bed was composed of a fluidyne engine constructed from copper pipe and plastic tubing coupled with a Fresnel lens for concentrating solar energy. The test bed was instrumented with temperature, pressure, and position sensors. The concentrated solar energy from the Fresnel lens provided ample power to operate the test bed, and tests were run in a wide variety of conditions. Temperature, pressure, and volume phasing along with indicated work are presented and discussed for operation of the engine with no externally imposed load.

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Introduction

This paper describes the development and characterization of a solar powered liquid piston Stirling engine test bed at the University of Colorado at Colorado Springs. The construction and operation of the test bed was intended to serve two purposes. First, it demonstrated direct solar-powered operation of a liquid piston engine with sunlight concentrated directly on the hot cylinder as opposed to using an intermediate heat exchanger. It was also used to characterize some aspects of the operational thermodynamic cycle of this class of engines. The test bed was only intended as a platform to explore feasibility and thermodynamic characteristics, not as a practical, power producing engine. Hence, electrical generation equipment was not included.

Liquid piston Stirling engines, sometimes referred to as “fluidyne” engines, use liquid pistons rather than the solid pistons found in conventional engines. Solid pistons normally require the design, manufacture, and maintenance of sliding mechanical seals. This not only requires relatively high tolerance manufacturing, it also necessitates periodic seal replacement thereby increasing maintenance costs over the life of the engine. Liquid piston engines do not require any mechanical seals which decreases costs. Liquid pistons also allow for unconventional piston and cylinder shapes which could be beneficial for some applications. Fluidyne engines

do not require the use of valves and advanced timing controls for engine operation. In fact, most fluidyne engines are simple devices that can be manufactured with relatively few tools and materials. This leads to very low capital costs and makes the fluidyne appealing for certain applications in which low-technology manufacturing, free or low-cost heat, and minimal maintenance are available [18]. While the fluidyne engine may have potential for some applications, the device's typical low thermal efficiency and poor specific power limit its use to situations with low-cost or free energy sources [3]. This includes, but is not limited to, pumping applications in developing countries or remote areas where frequent maintenance and attention may be unavailable.

One configuration for a fluidyne engine consists of a closed U-tube partially filled with liquid and connected at the bottom to a second tube also partially filled with liquid as shown in Fig. 1. The liquid partially filling the U-tube acts as a displacer piston which moves a working gas from a hot side heat exchanger to a cold side heat exchanger. When the gas in the working space shifts from side to side, it is heated and cooled causing the system pressure to increase and decrease. A second liquid column connects with bottom of the U-tube. In one possible configuration, the second column (known as the output column, tuning column, or tuning line) remains open to atmospheric pressure. As the system pressure rises and falls, the liquid in the output column moves up and down accordingly. This oscillatory motion provides the necessary feedback to force the movement in the displacer piston in the U-tube as well as providing one possibility for extracting work from the engine.

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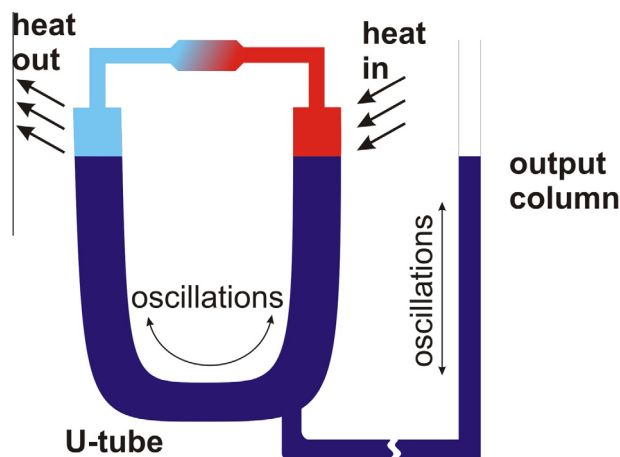


Fig. 1. U-tube fluidyne schematic.

The primary advantage of the fluidyne lies in the simple construction and operation of the device. A properly designed engine is self-starting and requires little maintenance. Some disadvantages of the fluidyne engine include typically low thermodynamic efficiency due to fluid friction, poor heat transfer characteristics, and the phase changes of the liquid columns that occur at high temperatures [18]. Also, virtually all working fluidyne configurations have relatively low power density [3].

The basic function and design, as well as many applications of fluidyne engines have been studied and reported in the literature. In a comprehensive book, West [20] described the history, scope and state of the art of fluidyne engine design. Fauvel et al. [3], explored the mechanism by which the tuning line supplied feedback to the displacer piston either via hydrodynamic coupling, hydrostatic coupling, or a combination of both. In the experimental study, Fauvel et al. concluded that a plain ended termination point yielded the best results with a phase lag of 49°, suggesting that the coupling mechanism from this type of fluidyne was a nearly exactly equal contribution of hydrostatic and hydrodynamic coupling. Fauvel et al. [4] addressed the response of both the displacer piston and tuning line to a varying system pressure. Fauvel and West [1] corroborated experimental data from Fauvel's previous experiment with analytical data obtained from differential equations describing the piston motion. Stammers [17] derived approximate algebraic expressions for minimum temperature differentials and for stability (beginning of oscillations) from thermodynamic theory, and verified the expressions in part with experimental measurements. All work was done for dry (evaporation suppressed) fluidyne configurations.

In applications of fluidyne systems, a comprehensive review by Wong and Sumathy [21] of water pumping systems driven by solar energy includes a description of fluidyne-type systems. This review article clearly shows the potential application of fluidyne engines for pumping water in the wider context of such systems. Van de Ven [19] considered the benefits of using a liquid piston Stirling engine as a mobile hydraulic power supply option. The efficiency and cost benefits of the liquid piston and Stirling engine design were examined. Applications to automotive power plants, robotics, construction equipment, irrigation pumping in developing nations and heat pumps were considered.

Fauvel and Yu [2] specifically addressed the design and effectiveness of a barrel-type fluidyne. This study focused on a practical fluidyne design. Orda and Mahkamov [12] described an empirical study of different water pumps, collecting efficiency and pumping capacity data. The study involved three different prototypes; the first employed the use of a heated element in a concentric cylinder

design, the second utilized flat-plate solar collectors in a U-tube design, and the third design improved the second design with a more compact form. All three fluidyne models contained a solid displacer piston in between the hot and cold side columns which was attached to a spring to aid in cycling the liquid back and forth. While Orda and Mahkamov only presented experimental data from their own fluidyne designs, their study presented physical data of solar-heated fluidyne engines which can operate under the varying conditions of solar collection and pumping needs.

Slavin et al. [14] described the development of fluidyne engines for medium-sized electricity generation application. They conducted a computer simulation which analyzed a complex fluidyne-type device that contained multiple valves and chambers. Özdemir and Özgüç [13] attempted to formulate a mathematical model based on hydrodynamic and thermodynamic equations and compared that model to an actual experiment. Similar to the Slavin et al. study, the model was built on hydrodynamic and thermodynamic equations, except for the additional interfacial conditions that occur in a wet fluidyne (evaporation and condensation). The simulation data was supported by data from experiment, however the authors noted that the weakness of the model was the modeling of the interfacial mechanisms.

Solanki et al. [15] and Markides and Smith [11] used an electrical analogy to create a complete linear model of an oscillating liquid piston engine which bears some similarity to the fluidyne engines of this study. Even though liquid inertia was nominally not a dominant factor in the operation of the engines under consideration in their study, it was found to have significant effect on the validity of the model. The phasing of the fluidyne engine, in contrast, depends principally on the inertia of the liquid pistons. Markides et al. [10] incorporated inertia and heat transfer nonlinearities into the analysis of this engine and found that this resulted in a markedly improved model. Both the linear and non-linear models predicted similar oscillation frequencies, but the non-linear model was able to account for stability and exergetic efficiencies. Solanki et al. [16] developed a linear heat transfer model that allowed for dynamic storage and release of heat in the heat exchanger combined with a dissipative thermal loss parameter. This model provided more accurate predictions of both operation frequency and efficiency. Markides and Gupta [9] implemented their oscillating liquid piston engine in a hot water circulation system reporting maximum flow rate of 850 L/h at zero head, and stalling near a pressure head of 8.4 m of water.

Kongtragool and Wongwises [5] performed a comprehensive review of Stirling engines powered by solar heat and low temperature differences. They didn't specifically examine liquid piston Stirling engines, but concluded that solar powered Stirling engines had potential for significant useful power generation applications. Kongtragool and Wongwises [6] also reviewed various power output prediction formulae for Stirling engines with particular attention to low temperature difference applications. They found that the Beale number correlations were the simplest and that a variation which they termed a "mean pressure power formula" would give the best results.

Kongtragool and Wongwises reported two studies [7,8] on Ring-bomb-type low temperature difference engines, one with two power pistons and one with four power pistons with atmospheric air as the working fluid. High intensity lamps were used as a heat source to simulate concentrated solar power. Tests were conducted at temperature differences between about 90 °C and 130 °C with the cold side temperature always maintained at 34 °C. Thermal efficiencies between 0.35% and 0.65% were achieved with dimensional Beale numbers between 1.3 and $2.5 \times 10^{-3} \text{ W}/(\text{bar cm}^3 \text{ Hz})$. Both engines operated at speeds in the neighborhood of 20–50 rpm.

While considerable development work has been accomplished with liquid piston Stirling engines, and a variety of similar engines

have been explored with solar and other power sources, quantitative measurements of system operational parameters (temperature, pressure, and volume) have not been reported for fluidyne engines powered with direct, concentrated solar energy.

System description

The test bed used a 50" by 37" (127 cm × 94 cm) Fresnel lens to concentrate sunlight onto the hot side of the engine. The displacer piston was constructed of 1" (2.54 cm) diameter piping and the tuning column consisted of ½" (1.27 cm) tubing. The curved portion of the displacer piston was made from 1" (2.54 cm) PVC pipe and connected to 1" (2.54 cm) copper pipe used for the vertical sections on both the hot and cold sides. The hot and cold sides of the displacer piston were connected with 3/8" (0.95 cm) plastic tubing to seal the working space. The tuning column was connected to the displacer piston directly below the hot side vertical section via a copper and CPVC insert. This joint was then connected to ½" vinyl tubing.

The framed Fresnel lens was mounted on a wooden stand which allowed the lens to remain focused on the fluidyne hot side as the lens was moved to different angles in order to accommodate movement of the sun through the duration of a test run. The stand supporting the Fresnel lens allowed both azimuthal adjustment by rotation of the entire wheel-mounted stand, and altitude adjustment by pivoting the rigid lens frame and extension arm around bolts mounted in a vertical structural member in line with the Fluidyne. This design enabled the focal point of the lens to remain at a fixed point as altitude adjustments were made.

Figs. 2 and 3 show the fluidyne engine and lens stand respectively. Table 1 lists significant system dimensions.



Fig. 3. Fresnel lens and stand.

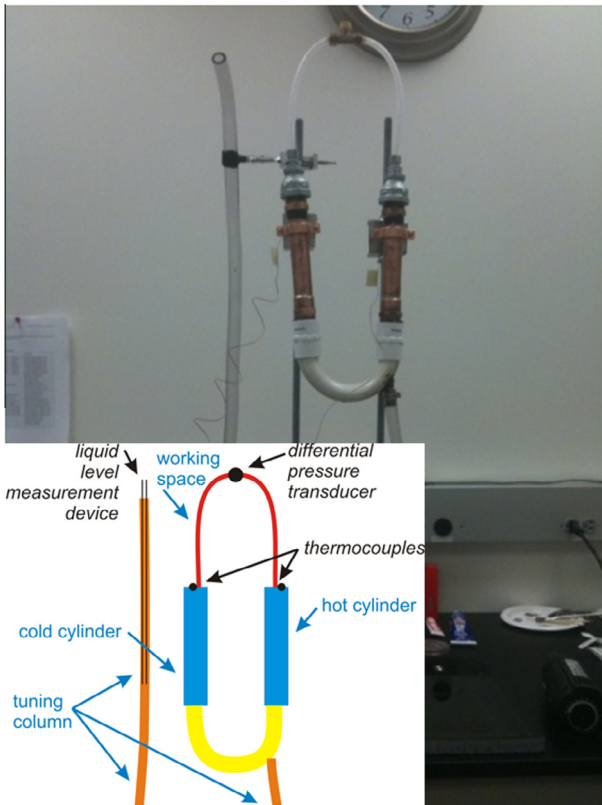


Fig. 2. Instrumented fluidyne engine.

Table 1

System dimensions.

Length of displacer piston	17.5" (44.45 cm)
Diameter of tuning column	0.5" (1.27 cm)
Diameter of tubing connecting hot and cold sides of the working space	0.25" (0.62 cm)
Length of tubing connecting hot and cold sides of the working space	15.5" (39.37 cm)
Length of tuning column	5.3' (161.5 cm)
Nominal system frequency	1.06 Hz

Instrumentation

The test bed was instrumented with two thermocouples, a differential pressure transducer, a liquid level measurement device, and a data acquisition system. The data acquisition system consisted of a Gateway Solo Pro 9300 computer running LabView with a Keithley KPCMCIA-16AIAOH data acquisition card. The position of the various instruments is diagrammed in Fig. 4.

The two type T 24 gauge thermocouples were manufactured by Omega Engineering, Inc. The thermocouple beads were approximately 1 mm in diameter. The thermocouples were placed at the top of the main copper cylinders out of reach of the oscillating displacer piston. These thermocouples recorded the air temperatures in both the hot and cold cylinders.

The differential pressure transducer, catalog number PX139-005D4V from Omega Engineering, Inc., measured the pressure inside the working space relative to the laboratory atmospheric pressure. The high speed of sound relative to the operating speed of the engine justifies the assumption that the pressure was always in quasi-equilibrium across the working space. The pressure

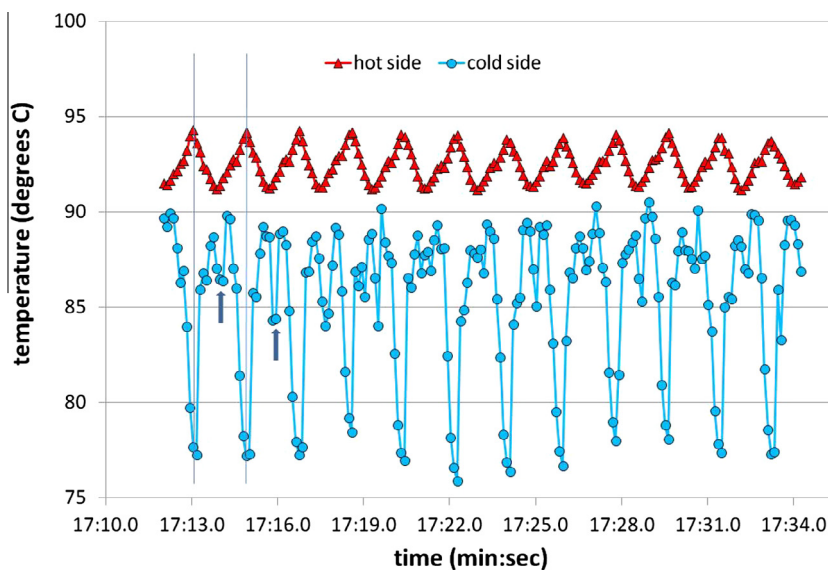


Fig. 4. Measured temperatures.

measurements were used to determine the pressure oscillation amplitude and phase and were also used to determine the indicated work of the engine.

The liquid level measurement device was fabricated from two sections of 1/16" (0.16 cm) diameter welding rod (ER70S-2) positioned parallel to one another in the tuning column. The electrical resistance between the rods was proportional to the water level in the tuning column. The rods were wired in parallel with a 510 Ω resistor. An additional 83 Ω resistor was wired in series with the parallel circuit to limit current to 15 milliamps from the 5 V power supply. The system was re-calibrated each day before testing in order to account for any change in conductivity of the water. The measurement of the tuning column height helped determine the phasing of the liquid pistons as well as the working space volume during engine operation.

The sampling rate was limited by the capabilities of the software and hardware. With all instrumentation in place, the data acquisition system could sample all sensors every 120 ms.

Measurement uncertainty

For the two thermocouples, the differential pressure transducer, and the level measurement device, the random and systematic errors used to determine the measurement uncertainty were estimated by operating the instrumentation at steady laboratory conditions. At a steady temperature, the temperature fluctuations of the hot side and cold side temperatures had standard deviations of 0.0325 $^{\circ}\text{C}$ and 0.0297 $^{\circ}\text{C}$. The system bias or systematic error was determined to be 0.1 $^{\circ}\text{C}$. Combining these in the standard root-mean-square way resulted in a total uncertainty for the thermocouple readings of ± 0.1 $^{\circ}\text{C}$. An identical procedure for the pressure measurements resulted in a total uncertainty of 0.010 PSI (69 Pa). The time responses of the temperature and pressure sensors were estimated to be 0.2 s and 0.1 s, respectively.

The tuning column line had to be calibrated daily using recorded voltage values across the electrodes when the liquid level was steady. The random error in the measured voltage corresponded to changes in liquid level of 0.011" (0.03 cm).

Using the root-mean-square approach, the total tuning column height uncertainty was calculated to be ± 1.81 " (4.6 cm). The tuning column height was also used to determine the working space

volume. This tuning column height uncertainty resulted in a working space volume uncertainty of ± 0.356 in³ (5.8 cm³).

Results

Testing took place at 38° 53' 30" N, 104° 48' 14" W. Weather conditions fluctuated during the testing period including various ambient temperatures, wind, and cloud cover conditions. While these conditions affected the output of the experiments, they are the same parameters that would affect a solar power fluidyne engine intended for practical use.

One of the primary goals of this project was to demonstrate the practicality of a solar powered fluidyne engine. Using a very rough estimate of the incident solar flux of approximately 900 W/m², the 1.2 m² lens might provide approximately 1100 W of power. Comparative preliminary testing between the Fresnel lens on a clear, sunny day and a 1000 W heat gun indicated that the radiative heat transfer from the lens was a much more effective heat transfer mechanism than the forced convection from the heat gun. For the system as designed, the Fresnel lens provided a more-than-adequate source of heat. Since fluidyne engines typically operate with very low thermal efficiencies, most of the heat input into the hot side of the system is also rejected from the cold side. When heat input came from the Fresnel lens, the experiment had to be temporarily halted after a time or else slowly rising temperatures could melt some parts of the device. Thus, the system never operated at a true steady-state, however, the change was slow enough that quasi-steady performance data could be gathered over sustained periods as long as twenty minutes.

Fig. 4 shows about 20 s of temperature data measured by the hot side and cold side thermocouples. This particular section was selected as typical of quasi-steady state operation in many different tests and under many different conditions. The amount of data shown in this and other figures is limited simply for the purpose of enhancing clarity and visibility. The temperature fluctuations over a cycle were approximately 4 $^{\circ}\text{C}$ on the hot side and 12 $^{\circ}\text{C}$ on the cold side. As might be expected since the same air is being pushed across both thermocouples, the hot and cold side temperature fluctuations were exactly 180 degrees out of phase with one another. Vertical lines are shown on the first two cycles to emphasize this visually, however, this and other phasing data that will be described later were derived rigorously by analyzing the data over

many cycles. While the hot side temperature fluctuations proceed more-or-less monotonically through heating up and cooling off around a cycle, it can be seen in Fig. 4 that the cold side temperature has a very short secondary dip during the warm part of the cycle. Two of these dips are indicated by arrows in Fig. 4 for the first two cycles. It is believed that the difference in behavior of the two measured temperatures is a result of the strong imposed heating being applied to the air on the hot side compared to the less effective combined convection and conduction cooling on the cold side. The cold-side thermocouple is first heated by air coming from the hot side of the device, then has a short time to cool off as the air flow slows to a stop and reverses, and then is reheated by the last hot air as it now passes in the opposite direction, and finally is cooled by the air that has had more time to cool off on the cold side of the machine.

Figs. 5 and 6 show the phasing between the temperature of the hot side thermocouple and the pressure (Fig. 5) and volume (Fig. 6). The data cover the identical time period as Fig. 4 for comparison. The pressure and hot side temperature are exactly in phase with each other. Note that the hot side temperature reaches a peak at the *end* of the flow of the working fluid from the hot side

toward the cold side of the device. A decrease in hot side thermocouple reading indicates that the flow is reversed, and is crossing the hot side thermocouple from the cold side of the device. That is, the peak in hot side temperature coincides with the maximum amount of air on the cold side of the engine. That this is also exactly coincident with the maximum pressure in each cycle indicates that the air temperature fluctuations are directly driving the engine motion. This result might be anticipated since all measurements in this study were conducted on the engine with no imposed external load. As a consequence, all indicated work output of the engine was used to overcome internal friction. Under such conditions, with the engine essentially running at idle, the air pressure varies inversely with the bulk average temperature of the air. In addition, the phase alignment is indicative of the very small (nearly negligible) effects of compressibility in this unloaded engine.

For the purposes of comparison, the volume of the working space is shown in Fig. 6 with the same temperature data as Fig. 5. It is clear that the minimum in the hot side thermocouple temperature (corresponding to the point when the maximum amount of working fluid is on the hot side of the engine) is close to, but not precisely coincident with, the maximum volume of

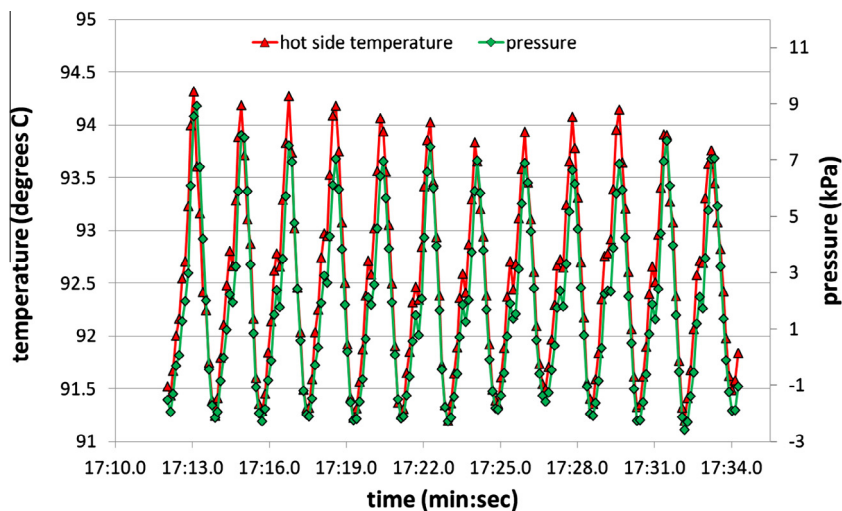


Fig. 5. Temperature and pressure phasing.

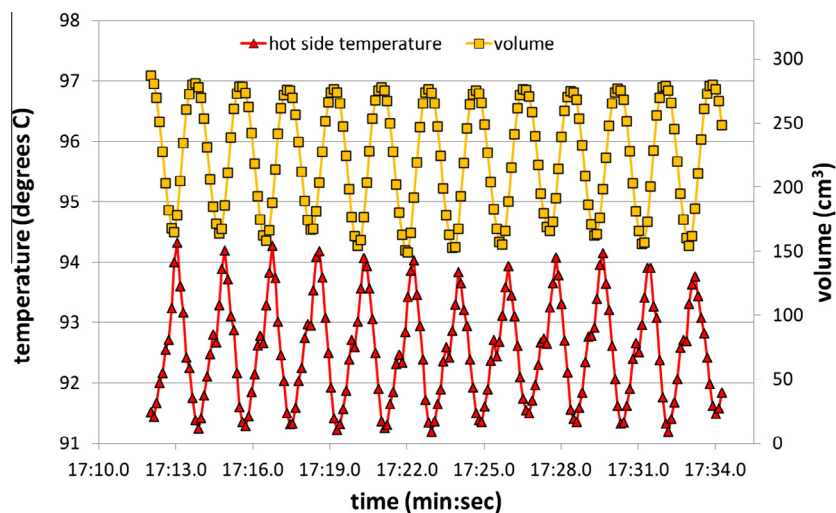


Fig. 6. Temperature and volume phasing.

the working space. This simply emphasizes the point made in the preceding paragraph that the pressure and volume fluctuations are driven directly by the average bulk temperature of the air.

Figs. 7 and 8 address the phase shift between pressure and volume (Fig. 7) and the corresponding indicated cycle work (Fig. 8). The pressure fluctuations are shifted by 108 degrees relative to the volume fluctuations. This shift, created by the combination of the fluid motions in the displacer and output columns, enables the engine to produce net work. As mentioned previously, in this study the engine was operated with no external load, so all indicated work went to overcome frictional losses of the liquid pistons moving with relative to the pipe and tubing of the engine. It can be observed that the gauge pressure drops below zero in each cycle indicating a negative gauge pressure in the working space.

Fig. 8 is an indicator diagram of the working engine for the same time period that has been used in all of the other plots. On this plot, the area enclosed by each cycle is proportional to the indicated work of the engine. Since no external load is present, and the cycle

period is dictated by the dimensions of the engine, increases in heat input resulted in larger fluctuations in the liquid piston motions. Since fluidyne engines function with a resonant feedback from the tuning line, the engine frequency is fixed by the geometry of the tuning line (output column) and the displacer (U-tube). The result is that unlike typical IC engines which are physically constrained to operate with the same displacement but can vary their speed, the fluidyne engine operates with constant “speed” but can vary the displacement. The result of larger liquid piston movements is higher friction resulting in larger enclosed areas on the indicator diagram. As an illustration of this, also included on Fig. 8 is an indicator diagram for a completely separate set of data taken on a different day with smaller liquid piston movements. It follows the same general trends, but the area enclosed by the indicator diagram is smaller.

Fig. 9 shows only the first four cycles of Fig. 8 plotted with different symbols in order to make the details of the cycle work more visible. Also overlaid in Fig. 9, shown with a heavier line and no

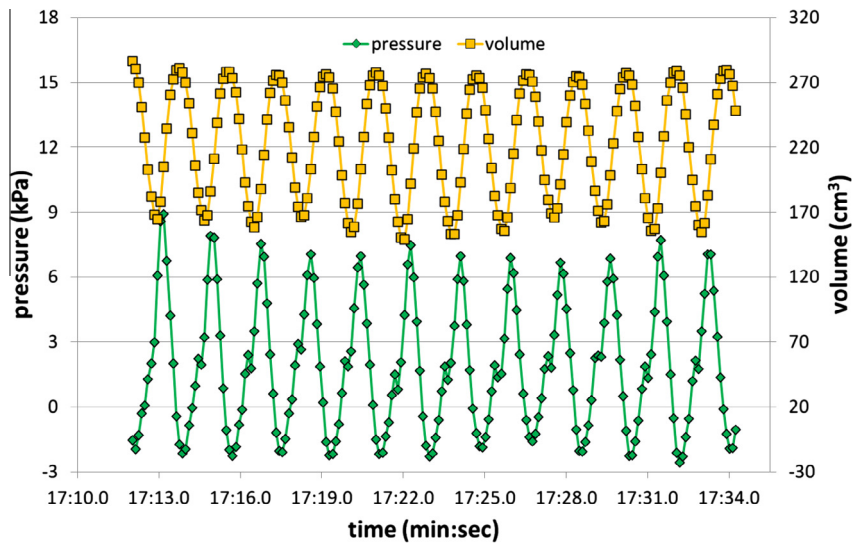


Fig. 7. Pressure/volume fluctuations and phasing.

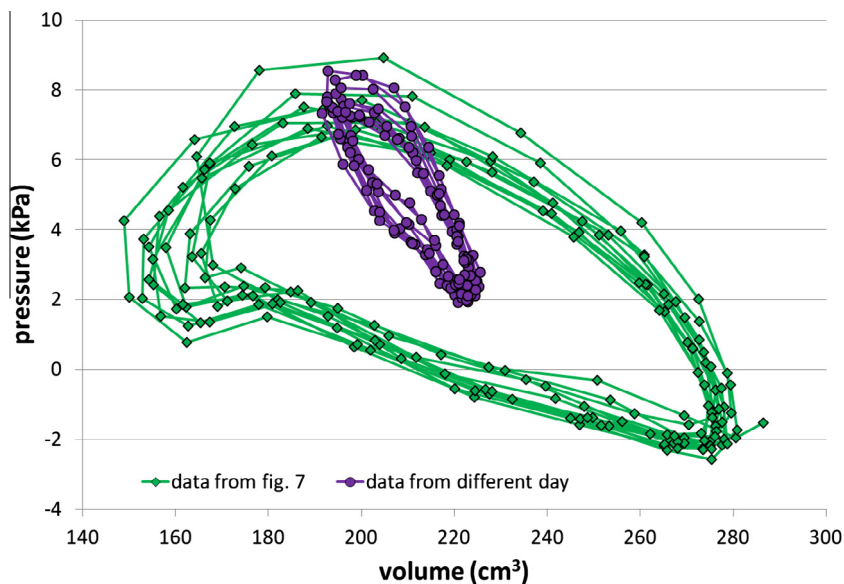


Fig. 8. P–V plot over multiple cycles.

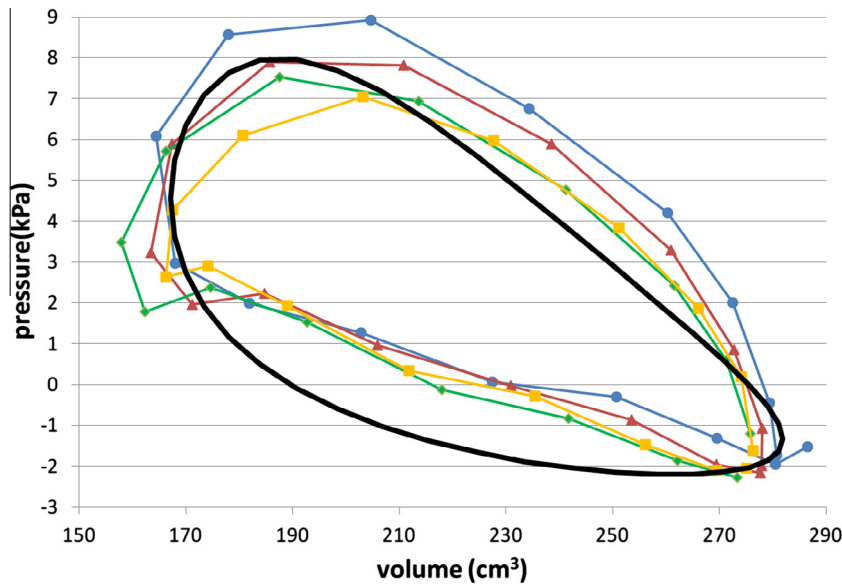


Fig. 9. P–V plot detail.

symbols, is a simple P–V prediction based on sinusoidal motion of the displacer piston and output piston with 90 degree phase difference, between the two and scaled to fit the measured volume and pressure fluctuations. The overall agreement is forced by the scaling, however, this scaling allows a comparison between the shapes of the measured cycle and the idealized P–V shape. It can be seen that the measured data has a more concave shape during compression and a more convex shape during expansion than the idealized plot. This is explained by noting that it can be seen clearly in Fig. 7 that the volume and pressure fluctuations deviate significantly from purely sinusoidal motion. Another difference is that the experimental phase difference between the displacer and output pistons is 72 degrees (corresponding to the 108 degree phase difference between pressure and volume fluctuations) rather than the 90 degrees assumed for the idealized case.

Fig. 10 explicitly shows the work of the engine for each discretely measured point around the cycles for the same data that has been used in the other plots. This work was calculated by multiplying the average pressure between each two data points with the volume

change between those points. Arbitrary cycle breaks are shown with vertical lines for the first three cycles in order to give context to the total net cycle work, indicated with heavy black lines for each cycle, and the work ratio (ratio of negative work to positive work during a cycle), indicated with lighter weight red lines. For the period shown, the work ratio averaged about 12%.

The Beale number, typically defined as the ratio of work output to the product of pressure, displaced volume, and frequency, is often used as a rough design parameter as well as an operational characteristic of a Stirling engine. For the device described in this study, the Beale number is always identically zero since all operation occurred with no external load. Nevertheless, for comparison with other engines operating at idle, a form of the Beale number can be calculated based on the indicated work as shown in Figs. 8–10. The dimensionless Beale number so calculated averages 0.44 over the data in those figures with a range between 0.40 and 0.50. While this is enormous compared with typical values (0.1–0.2), a direct comparison is not appropriate since this definition is using indicated, rather than brake work of the engine.

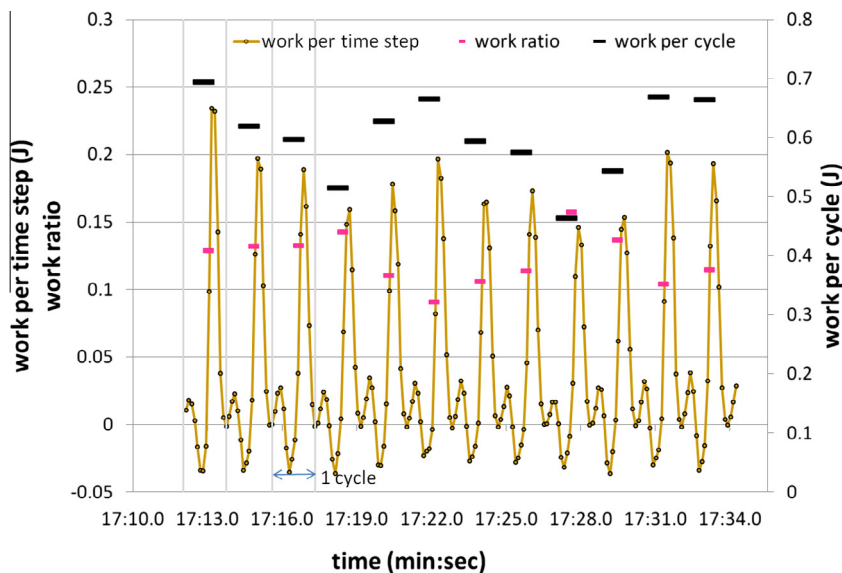


Fig. 10. Work.

Conclusion

The instrumented and directly solar-powered fluidyne test bed demonstrated consistent and repeatable operation. The Fresnel lens used provided ample energy to power the fluidyne. The unloaded engine used all indicated work to overcome internal friction of the moving liquid pistons. Operating in this state, the pressure and volume phase difference was 72 degrees compared to the idealized sinusoidal motions separated by 90 degrees. The dynamic mechanical response showed no delay from the thermodynamic expansion due to temperature changes of the working fluid. The constant speed nature of the fluidyne engine means that increases in heat input result in greater piston displacement for an unloaded engine. The ratio of negative to positive work over a cycle averaged 12% for operation of this engine with no external load.

This fluidyne test bed successfully demonstrated direct solar operation by focusing sunlight on the hot cylinder with a Fresnel lens. The unloaded engine compared favorably in operational characteristics with previous laboratory fluidyne models and shows that a functional solar powered application for fluidyne engines such as pumping water or generating a modest amount of electricity would be feasible. The remaining untested areas include development for long-term steady state operation and development for operation under an external load, however, these are modest additional steps. Laboratory testing on dynamometer loaded (non-solar powered) fluidyne engines shows that the external load tends to stabilize the dynamic operation of the engine. Simply implementing a higher capacity heat exchanger on the cold cylinder would serve to stabilize the thermal operations.

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